

ACHIEVING CONSISTENT MAXIMUM BRAKE TORQUE WITH VARIED INJECTION TIMING IN A DI DIESEL ENGINE

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ABSTRACT

Achieving Consistent Maximum Brake Torque with Varied Injection Timing in a DI Diesel Engine. (May 2014)

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The brake torque of a direct-injection diesel engine is known to plateau over a range of injection timings. Injection timing affects the engine's ignition delay and the fractions of fuel which burn in premixed and diffusion modes. Therefore, the characteristics of combustion for swept injection timings along the maximum brake torque plateau are determined. The research is conducted by varying injection timing at constant engine speed and load while measuring engine emissions and in-cylinder pressure, revealing the premixed and diffusion burn fractions as well as important engine and exhaust design criteria such as maximum in-cylinder pressure and exhaust composition. These results are significant in diesel engine design because cheaper, lighter engines may be built if desired maximum brake torque may be produced with lower peak in-cylinder pressures. More critically, different emissions composition profiles exist for the desired maximum brake torque, which will allow designers to choose injection calibrations which favor some emissions over others. In fact, both favorable emissions and in-cylinder pressure criteria can be achieved by implementing more retarded timings along the brake torque plateau without losses in efficiency or power output.

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NOMENCLATURE

°CA	Degrees Crank Angle
ATDC	After Top Dead Center
BMEP	Brake Mean Effective Pressure
BTDC	Before Top Dead Center
DI	Direct Injection
EGR	Exhaust Gas Recirculation
IMEP	Indicated Mean Effective Pressure
MBT	Maximum Brake Torque
MPRR	Maximum Pressure Rise Rate
LTC	Low Temperature Combustion
ROHR	Rate of Heat Release
RPM	Revolutions Per Minute
TDC	Top Dead Center

CHAPTER I

INTRODUCTION

In direct injection diesel engines, the fuel is injected into the cylinder without prior mixing with air. This injection begins before the piston has reached TDC, such that there is a delay period before the fuel and air are compressed to the point of autoignition, which is the compression-induced spontaneous combustion that defines diesel engines. Autoignition requires both some mixing of fuel and air and an increase in temperature and pressure which allows the mixture to combust. However, autoignition is achieved before all of the fuel has even been injected, so that the delay period is necessarily not long enough for all of the fuel to mix with air. Therefore, the first part of the combustion process burns the fraction of the fuel that has mixed with enough air to combust, but the combustion during the rest of the power stroke occurs by the diffusion of the rest of the fuel and air as the remaining fuel evaporates and spreads through the combustion chamber. These two types of combustion present in diesel engines are visible in photographs such as those taken in special visualization engines [1]. In light of these two burning modes, it is easy to see why changing the injection advance (how early injection begins BTDC) alters the ignition delay and allows a different fraction of the fuel to mix with air before combustion begins.

Because of the influence of injection timing on these two combustion mechanisms, some injection timings yield near-constant torque for a given engine speed. This plateau in the brake torque of DI diesel engines has long been known to exist. However, literature review has shown that the premixed and diffusion burn fractions present along this plateau have not been

catalogued, and means of using the torque plateau to achieve design goals have not been fully explored. Knowledge of the variation in premixed and diffusion burn for different injection timings is important for understanding the empirical evidence of near-constant MBT over a range of ignition delays. Then, by understanding the in-cylinder conditions responsible for the plateau, researchers may improve engine weight or reduce harmful emissions.

Hypothesis

The novelty of this topic required hypothetical connections between injection advance and MBT to be drawn from data presented in other ways. For instance, data sets in the literature often presented one operating parameter as a function of injection timing, but only loosely linked that to torque output. The hypothesis for the relationships of injection timing, premixed burn fraction, and diffusion burn fraction to MBT was derived from these trace links in existing data.

Heywood shows the region of very low slope (plateau) for BMEP vs. injection timing, which is useful because BMEP is actually a torque measurement normalized by cylinder volume [1].

While no explicit evidence is provided to demonstrate burn fractions during the plateau of BMEP, Heywood also illustrates increasing smoke emissions as injection timing is retarded along the plateau [1]. Since diffusion burn is known to produce more smoke than premixed burn, one would expect part of the plateau to be dominated by premixed burn (with characteristic low smoke emissions) for the earliest injection timings along the plateau. The plateau appears to be dominated by diffusion burn for the later injection timings along the BMEP plateau before the BMEP decreases more rapidly.

These smoke emissions characteristics indicate, therefore, that the upper plateau of brake torque (presented in this case as BMEP, which is proportional to torque divided by displaced cylinder volume) is indeed attained by both premixed and diffusion burn dominated combustion, as well as some balance of the two burn types for an injection timing in between the regions dominated by one type or the other.

Badami et al. present one useful set of data pertaining to cylinder pressures for varied injection timing [2]. This data includes both maximum cylinder pressure per cycle for different injection timings and IMEP for different injection timings. Since this study varied injection pressures and injection timing to determine the effect of injection pressure, not all injection pressures correspond to engine outputs near the MBT plateau. However, one injection pressures resulted in a region of low slope in the curve of MEP vs. injection timing. This corresponds to near constant torque. From this study, it appears that peak pressure can be minimized by choosing the latest possible injection timing while still producing maximal torque in the plateau.

To summarize, two injection timing bands of a few °CA appear to provide roughly equal brake torque while offering significantly different emissions profiles. Injection timings (within the MBT plateau) closer to TDC appear to yield more smoke, indicating dominant diffusion burn, and more advanced injection appears to yield more NO_x, indicating dominant premixed burn. Additionally, injection timings closer to TDC appear to apply lower mechanical stresses to the cylinder. This is consistent with the hypothesis that diffusion burn dominates in the later injection range of the MBT plateau, since premixed burn is the part of combustion which produces the greatest instantaneous pressures in the cylinder.

This research aims to confirm the above hypothesis, to determine the actual premixed and diffusion burn fractions present at the injection timings which attain MBT within the torque plateau, and to explore the practical implications of the differing premixed and diffusion burn fractions.

Implications

One goal of this study is to determine what fractions of premixed and diffusion burn can be achieved at various injection timings while still yielding MBT within the plateau region. This will allow selection of an injection timing which can minimize engine emissions while still attaining high torque output, since different burn compositions cause different emissions (namely, a tradeoff exists between NO_x for the premixed burn case and particulates for the diffusion burn case at conventional combustion temperatures). The other goal is to determine which injection timing can achieve the lowest maximum cylinder pressure per cycle while still achieving the same desired MBT from the plateau region. This will allow engines to undergo lower peak mechanical stresses without sacrificing performance and thus be built at lower weight and cost. Additionally, this study provides an opportunity to compare a common mass fraction burned integration method to a different, novel approximation based on modelling the fuel injection profile.

CHAPTER II

LITERATURE REVIEW

The diesel cycle has been studied continuously since its introduction in 1897, yet diesel combustion still presents new frontiers to modern engineers. Our understanding of the phenomena of diesel combustion has grown as technology provides deeper insights into in-cylinder processes, and today's engineers know so much about the general behavior of engines that they may explore specific, detailed topics, ranging from optimal turbocharger or piston geometries to unconventional diesel combustion modes. While the bulk of early research focused on increasing power output and engine speed, much of today's research focuses on increasing fuel efficiency or improving emissions, often through narrow avenues, such as the fuel injection spray cone angle. Even so, modern research on any specific diesel topic is grounded in appropriate phenomenological models and the current breadth of knowledge. Therefore, this review concludes with a brief summary of important implications for the relationship between ignition delay and torque which can be drawn from previous research.

Phenomenological Models

Since the original modeling of the diesel cycle as an internal combustion cycle in which combustion can be said to occur at a constant pressure, our knowledge of the specific behavior of fuel and air inside a diesel cylinder has vastly increased. Sir Harry Ricardo described the phenomena of combustion as they were understood in 1931, citing three stages of diesel combustion [3]. Early during the injection of the fuel, combustion is delayed while fuel and air mix because pressure and temperature do not yet thoroughly exceed the ignition point. Then, this

mixed fuel combusts very rapidly when the compression is great enough. Lastly, Ricardo posited combustion of the remaining fuel as each fuel element is injected into the now hot and pressurized cylinder. However, Ricardo's model came from the intuitive explanation to the limited, rough data available at the time, rather than from knowledge of the in-cylinder processes such as we have today.

The watershed in diesel phenomena was W.T. Lyn's study of burning rates in diesel engines, made possible by computers and improved in-cylinder sensors [4]. Lyn modeled combustion by representing elements of injected fuel as ready-to-burn mixtures with air over time. Basically, Lyn's model shows each sequential element of injected fuel mixing with air, thereby becoming available for combustion. However, as Ricardo noted, diesel combustion must always have a delay [3]. Therefore, Lyn's model rearranges a portion of the injected fuel which has mixed sufficiently to burn during the ignition delay (before the autoignition temperature and pressure are attained) by removing it from the delay period and adding an equivalent area to the rate of burning at the start of combustion [4]. This model predicts the behavior of the engine well if ignition delay is known, and it provides a useful approximation at both short and long ignition delays. The real benefit of this phenomenological model is that it provides a data-driven correlative representation of the phenomena Ricardo described. Lyn revealed a means for understanding the delay period, the very high initial rate of burning, and the long mixing-controlled burn by demonstrating the relationships among load, injection timing, ignition delay, and burning rate for a given engine configuration [4]. The correlation changed for different engines, such as direct versus indirect injection engines, but the model was revolutionary for its

insight. Lyn phenomenologically represented the delay, premixed combustion, and mixing-controlled combustion periods.

In spite of this insight, engineers would not be able to confidently describe the fuel-air interactions at each stage of combustion for decades. In the 1990s, Dr. John Dec used accessible cylinders and laser imaging to produce step-by-step images of injection and combustion, which provided much more detail to the three-stage phenomenology expressed by Ricardo and Lyn [5]. The images showed the onset of autoignition and, most importantly, clarified the interactions between fuel and air during the mixing-controlled or diffusion burn period. Dec shed light on several problems with the old models. For instance, Dec showed that the initial premixed burn is actually fuel-rich, not roughly stoichiometric as the old models often supposed [5]. This means that even during premixed burn, soot can be formed, and that therefore the soot formed during premixed burn is largely annihilated by other mechanisms during the combustion stroke.

Additionally, Dec's data revealed that premixed burn likely occurs over the entire volume of the mixed fuel and air, rather than at the periphery of the jet [5]. Dec's research also indicates that fuel undergoes burning in a slightly upstream standing pre-mixed flame before reaching the mixing-controlled flame at the jet's periphery [5]. Such revelations that couldn't have occurred without optically accessible engines and laser imaging are the most important impacts of Dec's update to the diesel combustion model. While it only covers combustion up to the end of injection, the model helps explain performance and emissions of direct injection diesel engines.

Current Topics in Diesel Research

From these conceptual milestones, modern diesel engine research covers a wide variety of disparate topics. While these topics do not address torque output as a function of injection timing in DI diesel engines, they all address contributing factors to the performance and efficiency of diesel combustion.

Direct and Indirect Injection

To preface this discussion of special topics in diesel research, it is important to differentiate between the two primary methods of establishing diesel combustion. In diesel engines with both injection modes, air alone is inducted through the intake valve or valves, and unmixed fuel is injected separately through an injector. In direct injection diesel engines, fuel is injected directly into the cylinder itself, and all combustion takes place in the cylinder. In indirect injection engines, by contrast, there is an additional chamber, often called the prechamber, into which the fuel is injected [1]. Then, autoignition conditions are easily achieved in the prechamber, initiating combustion outside of the cylinder main chamber [1].

Direct injection systems are useful in larger diesel engines, as well as in smaller engines in which air swirl is generated to improve mixing in the cylinder [1]. However, small, high-speed diesel engines often use indirect injection systems. Indirect injection offers better mixing, either through turbulence or swirl in the prechamber, and indirect injection allows for lower injection pressures [1]. Additionally, indirect injection engines typically generate higher BMEP than that of direct injection engines [1]. On the other hand, direct injection engines are always more

efficient than indirect injection engines of comparable size, with indirect injection usually yielding 15% higher best brake specific fuel consumption (bsfc) [1].

The following topics all cover research in direct injection (DI) diesel engines, but basic differences between the two systems provide important background to the discussion. Notably, indirect injection engines are less relevant to this topic because injection timing variations do not affect NO_x emissions in indirect injections nearly as strongly as they affect emissions in DI engines [1].

Low Temperature Combustion

One of today's popular research topics is low temperature combustion because it offers the opportunity to simultaneously reduce several emissions and to increase combustion efficiency. While researchers have long been aware that reducing combustion temperatures also reduces NO_x emissions, conventional combustion modes result in increased soot at lower temperatures. However, the breakthrough behind LTC is that exceptionally low temperatures (as far as combustion temperatures go) actually decrease net soot creation, as soot oxidation dominates soot formation [6].

Because the necessary temperatures are very low compared to typical combustion temperatures, LTC cannot be achieved without significant exhaust gas recirculation (EGR), which returns low-oxygen exhaust gases to the combustion chamber, delaying autoignition and allowing greater fuel and air mixing [7]. This premixed combustion is vital to simultaneously reducing NO_x and soot because diffusion burn is largely responsible for soot formation [7]. Additionally, high EGR

levels further reduce combustion temperatures, improving the ratio of soot oxidation to soot formation [6,7,8]. Another factor which assists in attaining LTC is that “the increase in ignition delay as timing is retarded from the location of minimum ignition delay is greater than that as timing is advanced from the same location” [6]. This means that marginal increases in the duration of fuel and air mixing are actually greater as injection timing is retarded from the point of minimum ignition delay, so that less advanced injection timings allow easier LTC. These less advanced timings do not cause fuel impingement on the cylinder walls, whereas the very advanced injection timings necessary to achieve the same level of mixing do indeed cause such impingement. Fuel that impinges on cylinder walls causes high unburned hydrocarbon, carbon monoxide, and soot emissions, which greatly reduces the usefulness of the LTC mode [6,9].

Besides being problematic at very early injection timings, fuel impingement is detrimental to LTC performance at high load, since greater fuel flow increases the amount of fuel which may escape the piston bowl and impinge on the walls. In an effort to improve LTC’s ability to handle more varied loads, some research has studied the effects of fuel ignitability and volatility on LTC load limits. While volatility plays a small part, ignitability can indeed alter the load limits of the LTC mode [10]. For fuels with a predicted cetane number (PCN) below 40, the load range becomes narrower, but LTC may be achieved at higher loads [10]. Further changes to the injection timing to maintain constant combustion phasing (defined as the crank angle at which 50% of fuel is burned) can further expand the load range while still offering higher load capability in the LTC mode [10].

The LTC mode not only reduces NO_x emissions, but it also enables regeneration of lean NO_x traps (LNTs) during rich combustion [7]. This occurs since rich LTC produces high CO and unburned hydrocarbon (HC) emissions, which recharge the LNT. However, such an operating condition causes higher fuel consumption, and the low temperatures involved prohibit catalytic elimination of the excess CO and HC emissions [7]. Although LTC has these emissions problems of its own, it remains popular because conventional diesel combustion is rapidly requiring more and more expensive exhaust aftertreatment systems to meet more stringent emissions standards [10].

Lastly, since the LTC mode can only be identified by expensive in-cylinder instrumentation, today's technology limits engineers' ability to identify whether or not LTC is actually occurring in a normal diesel engine (for instance, an automobile engine) [8,10]. Therefore, researchers have sought to determine whether or not ignition delay itself may be able to indicate the occurrence of LTC combustion in a typical diesel engine. Unfortunately, neither of two common methods of describing ignition delay allows reliable prediction of the LTC mode [8].

Injection Parameters

As the discussion of LTC hinted, changing injection parameters, such as fuel pressure, spray cone angle, and nozzle design, can greatly alter the performance and capabilities of DI diesel engines. As in the case of LTC, very advanced injection timings are often used to achieve premixed compression ignition and reduce NO_x emissions through mostly premixed combustion. Since conventional spray cone angles often cause fuel impingement on cylinder walls for advanced injection timings, for example, some research attempts to mitigate this problem

through the use of narrow spray cone angles [9]. Lechner et al. combined smaller injector holes and split injection to reduce fuel penetration (and thereby the likelihood of spray-wall interaction) [9]. With the goal of reduced NO_x through premixed combustion in mind, the study combined EGR and the small-angle spray cones at very early injection timings, eventually concluding that a very narrow spray angle of 60° provided the best reduction in NO_x, but cost more in terms of fuel consumption and smoke emissions [9].

Entirely beyond the scope of LTC and premixed combustion, however, is the fact that varied injection parameters enable dramatic improvements in performance during completely conventional combustion modes. One topic particularly relevant to the relationship between injection timing and MBT is the effect of injection pressure on DI diesel performance.

Essentially, increasing the injection pressure forces more fuel into the cylinder by the end of the ignition delay, resulting in more intense premixed burn [2]. Stronger premixed burn naturally causes greater peak pressure in the cylinder. This result in particular is useful for the research preceded by this review. Additionally, increased injection pressure greatly increases IMEP [2].

While a similar trend is observed for more advanced injection timings, the magnitude of this increase in IMEP is much greater for increased injection pressures than for more advanced injections [2]. In summary, Badami et al. reveal that increased injection pressure is a powerful tool for improving performance, and that the additional power required by stronger fuel pumps is offset by the larger increase in power produced by the engine with higher injection pressures [2].

In engines which utilize high injection pressures, one research frontier is cluster nozzles. A cluster nozzle is an injector nozzle in which several small injector holes are arranged in groups

around the injector tip. These clusters benefit from their smaller orifices while maintaining the advantages provided by conventional nozzle spatial distributions, since the clusters of small orifices are located where larger-diameter single orifices would normally be [11]. Using the higher injection pressures discussed above, DI diesel engines with cluster nozzles may attain reduced spray tip penetration, which helps avoid spray-wall interaction, and better fuel atomization, which improves mixing [11]. One downside of the cluster nozzle is the need for more EGR, since cluster nozzles cause higher combustion temperatures than conventional nozzles, resulting in high NO_x emissions [11].

Turbo- and Supercharging

While injector characteristics can improve fuel delivery, the necessary counterpart to improved injection strategies is improved air induction. Namely, turbocharging and supercharging are important fields for improving air density in diesel engines. Today, the majority of diesel engines are turbocharged. Turbocharging offers diesel engines the ability to burn more fuel (and deliver more power) by increasing the amount of air present for combustion [1]. While the output of naturally aspirated diesel engines is limited by maximum tolerable smoke emissions, the output of turbocharged diesels becomes constrained by the mechanical stresses and sometimes thermal loading in the components [1]. Therefore, a rather arbitrary constraint (smoke emissions) is replaced by a material constraint (engine strength), allowing boosted engines to achieve performance similar to significantly larger engines [1]. Past research has also shown that turbochargers in series can produce greater performance at lower cost than multistage single turbochargers [1].

This prior research serves as the backdrop for the limited additional turbocharger reading conducted in this literature review. Since the present study attempts to maximize torque while improving emissions or reducing engine weight, a recent study of turbocharging a small DI diesel engine offers insight into increasing torque and reducing emissions. Turbocharging increases an engine's peak power and torque, and it also improves specific fuel consumption and emissions [12]. Both waste gated and variable geometry turbochargers allow for improved performance, with the variable geometry turbocharger outperforming the waste gated turbocharger [12]. Lastly, turbocharging allows for slightly reduced engine noise since boosting reduces ignition delay and premixed combustion [12].

Additionally, variable valve train (VVT) systems (rather common in gasoline engines, but not seen in production diesel engines) manipulate air induction and exhaust processes to improve engine performance [13]. VVT systems can create pressure wave supercharging, which increases inducted air through tuned pulsation of pressures in the exhaust and intake ports [13].

Another benefit of VVT systems is improved air flow, which can increase swirl and fuel-air mixing [13]. In addition, VVT systems can change the effective compression ratio of a DI diesel engine, considerably changing the combustion behavior. In Tomoda et al.'s study for Toyota, the valve train could vary the effective compression ratio from a low of 12.5 to a high of 13.8, with a base cylinder compression ratio of 13.4 [13]. Lower effective compression ratio at high loads improves engine emissions but reduces volumetric efficiency, while higher effective compression ratio at low loads decreases unburned HC emissions, eases cold starting, and can reduce NO_x alongside increased EGR [13].

Implications for Maximum Brake Torque

Prior engine research is often merely tangential to the topic of torque output over varied injection timings, but the existing body of knowledge presents powerful implications for the present topic. The phenomenology of diesel combustion speaks for itself, demonstrating unavoidable, fundamental processes for any diesel engine research. Engine research pioneer Sir Harry Ricardo explained long ago that regardless of conditions prior to combustion, “a delay of some sort there must always be” [3], thereby establishing the simplest ground-rule for this particular study, which centers on the influence of delay upon performance and emissions. Additionally, Lyn and Dec together laid the groundwork for a correlative, data-driven understanding of premixed and diffusion combustion and the outcome of each combustion type [4,5]. These phenomena are inextricable from the study of the MBT plateau in DI diesel engines.

In terms of the hypothesis of the current study, prior research on low temperature combustion is vitally important. LTC rose to prominence as a research topic purely because of the desire to reduce multiple emissions simultaneously [6]. While dual reduction of NO_x and smoke emissions is not possible along the MBT plateau in the conventional diesel combustion mode, the LTC literature actually served to emphasize the importance of the smoke-NO_x tradeoff, revealing that the MBT plateau enables selection of an injection timing which minimizes the production whichever constituent is more difficult to trap. For instance, it is often more difficult to process NO_x than to physically filter out particulates in exhaust aftertreatments. Thus, despite the difference between the MBT plateau research and previous explorations of the LTC mode, the background provided by LTC research sparked a central idea motivating the current research.

The improvements produced at the frontiers of injection parameter research will likely not be particularly relevant to this study, but Badami et al. did invigorate a very attractive possibility. From data presented in their study, the opportunity to reduce peak pressures while achieving the desired torque was identified. One injection pressure included test points along part of the MBT plateau, demonstrating both consistent torque output and variable maximum in-cylinder pressure [2]. As hypothesized, the chance to reduce peak pressures is attractive because lighter engines may be constructed to compensate for reduced mechanical stress.

Similarly, most of the information on turbocharging is not fundamental to understanding the MBT plateau. Nevertheless, turbocharging is an important consideration in this case, since engine operating conditions were shown to vary tremendously between naturally aspirated and turbocharged states. Those who experiment with the MBT plateau must be cognizant of the potential for great differences between injection timings and engine performance characteristics along the plateau for naturally aspirated and turbocharged engines. Since most diesel engines are now turbocharged, the benefits of greater torque and lower fuel consumption [1,12] should be useful in studying the practical outcomes of the MBT plateau.

CHAPTER III

METHODS

This study uses a direct injection diesel engine to explore the brake torque curve of conventional combustion at varied injection timings. A high load condition is used to widen the torque plateau region and offer more viable data in terms of fuel burning conditions at each injection timing along the plateau.

Engine, Fuel, and Dynamometer

The engine used in this study is a four cylinder medium-duty diesel engine, which is equipped with a high pressure common rail fuel system to supply electronically controlled direct injection fuel injectors. The engine's specifications are listed in Table 1. The fuel used in the tests is standard Diesel #2, with properties given in Table 2.

Table 1: Specifications of the medium-duty engine tested in this study [16].

Bore	106 mm
Stroke	127 mm
Displacement	4.5 L
Rated Power	115 kW at 2400 rpm
Compression Ratio	16.57 ¹ (nominally 17:1)
Ignition	Compression
Fuel System	Electronic common rail, direct injection
Air System	Turbocharger with EGR

¹ Measured by oil displacement

Table 2: Summary of the properties of Diesel #2, the fuel used in this study [16].

Property [Standard]	Diesel #2²
Density (kg/m ³) [ASTM D4052s]	825.5
Net heat value (MJ/kg) [ASTM D240N]	43.008
Gross heat value (MJ/kg) [ASTM D240G]	45.853
Sulfur (ppm) [ASTM D5453]	5.3
Viscosity (cSt) [ASTM D445 40C]	2.247
Cetane Number [ASTM D613]	51.3
Hydrogen (%-mass) [SAE J1829]	13.41
Carbon (%-mass) [SAE J1829]	85.81
Oxygen (%-mass) [SAE J1829]	0.78
Initial boiling point (°C) [ASTM D1160]	173.4
Final boiling point (°C) [ASTM D1160]	340.5

To maintain the desired testing speed, the engine is coupled to a DC motoring dynamometer, which provides a torsional load to the engine through automatic feedback control of the inductor current.

Experimental Test Matrix

The study comprises test runs at 17 injection timings, with speed, EGR, load, and injection rail pressure held constant. All trials occur in the conventional combustion mode described in the phenomenological models, as opposed to the LTC mode discussed in Chapter II.

The fuel injection duration is held constant for each test condition with a constant rail pressure of 1000 bar, with a nominal fuel mass flow rate of 2.5 g/s. The EGR valve is closed for all conditions. The torque load imposed by the dyno is varied in order to maintain an engine speed

² Measured or calculated by Southwest Research Institute (San Antonio, Texas)

of 1400 rpm while varying injection timing. The injection timing sweep is performed while holding all other control settings constant.

Measurements and Data Acquisition

The most important measurements in this study are in-cylinder pressure and brake torque. The in-cylinder pressure is measured in cylinder #1 (the forward-most cylinder) by a piezo-electric pressure transducer with a 0.2°C resolution. The piezo-electric transducer is calibrated to ensure faithful measurement of in-cylinder pressure [14]. Pressure measurements are averaged over 300 consecutive cycles to remove cyclic variation and instead represent the steady state operating conditions studied. The brake torque is given by the dyno load cell. Additionally, fuel injector current and fuel injector needle motion are recorded to quantify the injection timings.

Other measurements used in the study include fuel flow rate (using a positive displacement meter), exhaust nitric oxide concentration (using a chemiluminescence technique), and exhaust smoke concentration (using a reflective smoke meter technique). The exhaust samples (tested for nitric oxide concentration and smoke concentration) are delivered to their respective analyzers through sample lines that are heated to 190°C.

Measurement instruments are calibrated to reduce systematic uncertainty [15], but random uncertainty in engine testing is often high because ambient conditions, including air temperature and humidity, affect test results. Therefore, the operating conditions of the first sweep of injection timing are repeated in the second sweep, to confirm the trends observed in the first test runs. However, in order to improve the resolution of torque in terms of injection timing,

additional operating points are added to the second sweep, better defining the torque plateau discussed in Chapters I and IV. For the trials performed on both days (the timings of the first injection sweep), data is averaged and statistically analyzed.

In some figures, for example in those showing exhaust species concentrations, lines connecting data points exist to distinguish different data series and not to suggest a trend between pairs of data points.

Calculations

The measured data are converted to other useful quantities via calculations, most of which stem from standard practices in engine research (such as the calculation of the apparent heat release rate, $\delta Q_{HR}/d\theta$). One novel calculation is the one employed to offer comparison against a more standard method of determining how much fuel was burned by the premixed and diffusion mechanisms.

Apparent Heat Release Rate

The apparent heat release rate is a common quantity in engine research and is derived from the First Law of Thermodynamics [16]. An energy balance accounting for energy transfers and storage in the cylinder is shown below [Equation 1], and the First Law of Thermodynamics dictates that the rate of heat addition into the control volume minus the rate of work out of the control volume is equal to the rate of change of the cylinder's internal energy [16]. The heat terms (δQ) are positive when heat is entering the control volume, either from combustion reactions or through heat transfer across the cylinder walls. The work term is positive when work

is being done by the cylinder contents on the piston (and thus leaving the control volume, accounted for by the negative sign in front of this term). The internal energy term is positive when the internal energy of the cylinder contents is increasing (which is usually sensed as an increase in temperature).

$$\delta Q_{HR} + \delta Q_{HT} - \delta W = dU_{CV} \quad (1)$$

Since the combustion process is a complicated chemical reaction, its kinetics are ignored in this simplified calculation, and combustion is instead modelled as positive heat transfer into the cylinder, adding thermal energy to the cylinder contents. This simplification is the reason that the solution is termed *apparent* heat release rate.

The work term is calculated from the known cylinder dimensions and the measured in-cylinder pressure. In Equation 2, the term dV represents the rate of change of the cylinder volume, determined from the cylinder dimensions.

$$\delta W = p * dV \quad (2)$$

In general, the change in internal energy accounts for changes in the temperature of the cylinder contents. For simplicity, the contents are modeled as an ideal gas mixture, which enables energy calculations based solely on temperature. Each species' contributions to the change in internal energy may be added together because of the ideal gas mixture assumption. In Equation 3, x_i represents the species mass fraction, m is the total trapped mass, C_v is the constant volume specific heat of the species, and dT is the rate of change of the temperature of the contents.

$$dU_{CV} = \sum_{species,i} x_i m C_{v,i} dT \quad (3)$$

The temperature change dT is calculated by applying the ideal gas law, using the measured pressure, cylinder geometry, known mass, and gas constant. The species concentrations and their specific heats are calculated using the JANAF tables [17].

The Hohenberg correlation allows calculation of the heat transfer, assuming that the cylinder walls are at a constant temperature and that radiation exchange is negligible [18]. Because tests are conducted at steady conditions, cylinder wall temperature is assumed constant with respect to time, and in diesel combustion, radiation is negligible as a mode of heat transfer. In Equation 4, the temperature difference (ΔT) represents the difference between wall temperature and the temperature of the cylinder contents, such that heat transfer out of the cylinder is negative by the convention stipulated in Equation 1.

$$\delta Q_{HT} = mAhd(\Delta T) \quad (4)$$

This method is validated by Foster, and by Depcik et al. [19,20].

Fractions of Fuel Burned by Premixed and Diffusion Mechanisms

There is no universal standard for determining the fractions of total heat released by the premixed and diffusion burn modes, but one simple and commonly used method is as follows.

The heat release rate is integrated over the crank angle in two parts. The first integral is bounded by the beginning of the positive region of the heat release rate profile and the local minimum after the largest heat release rate peak, and the second integral is bounded by that same local minimum and the nominal end of the heat release rate profile.

Added together, these two integrals represent the total energy released by the burning fuel. The first integral divided by the total corresponds to the fraction of energy released by premixed combustion, whereas the second integral divided by the total represents the fraction of energy released by diffusion combustion. Since the total energy released corresponds to the energy released by the fuel being burned, these fractions demonstrate the amount of fuel burned by each of the two mechanisms of interest.

A Novel Method to Estimate Premixed and Diffusion Burn Fractions

Another way to account for the premixed and diffusion burn fractions of the fuel combusted each cycle comes from modelling the injection profile. Using a model of the injection profile with respect to time on a degree-crank-angle basis, one can approximate the amounts of fuel injected before and after ignition. Then, for simplicity, all fuel injected during the ignition delay is said to burn in the premixed mode, and all fuel injected thereafter is said to burn diffusively. Therefore, this method offers an alternative to the more standard heat release integration method, and provides an interesting point of comparison against that standard.

The model of the fuel injection profile used was created for a technical paper by Bittle and Jacobs [21], and it is used to draw conclusions about the effectiveness of such a novel method compared to the common approach of heat release integration, as well as to facilitate discussion of uncertainty and error in each of the two means of determining these fractions.

The model simplifies the phenomena experienced by the fuel stream as the needle lifts (and is seated again) by assuming a linear increase in the fuel flow rate followed by a linear decrease,

forming an isosceles triangular fuel injection rate profile [21]. In reality, a shape with a plateau (such as a trapezoid) is likely to represent the fuel injection rate profile, since the needle will achieve full lift and a quasi-steady flow should be achieved before the needle valve reseats itself. However, the simplified triangular model allows rapid, efficient computation for many parameters beyond fuel flow rate, and it is easily integrated. For instance, the trapezoidal numerical integration performed in this study introduces no additional error into the triangular fuel flow model.

When implementing this integration to determine the premixed and diffusion burn fractions, the same nominal start of combustion used in the ROHR integration (the time in $^{\circ}\text{CA}$ at which the ROHR becomes positive) will be used as the cutoff point for the premixed fuel. Therefore, the premixed integral will be evaluated from the beginning of injection until the nominal start of combustion and the diffusion burn fraction will follow from that nominal start of combustion to the end of injection.

CHAPTER IV

RESULTS

At the high load condition studied, the brake torque curve spans approximately 15 °CA on an injection timing scale, from 20 °BTDC to 5 °BTDC. The high load condition enables a wider torque plateau, which in turn allows greater resolution in terms of quantifying the mass fractions of fuel burned in both the premixed and diffusion modes. The higher resolution was achieved by introducing new injection timings during the second test run [Figure 1]. Since several of the injection timings were tested only once, but occur in between the other timings which were tested twice, the average deviation in some parameters (such as NO_x concentrations) at the repeated timings is used to approximate the uncertainty in those parameters at the single-trial timings.

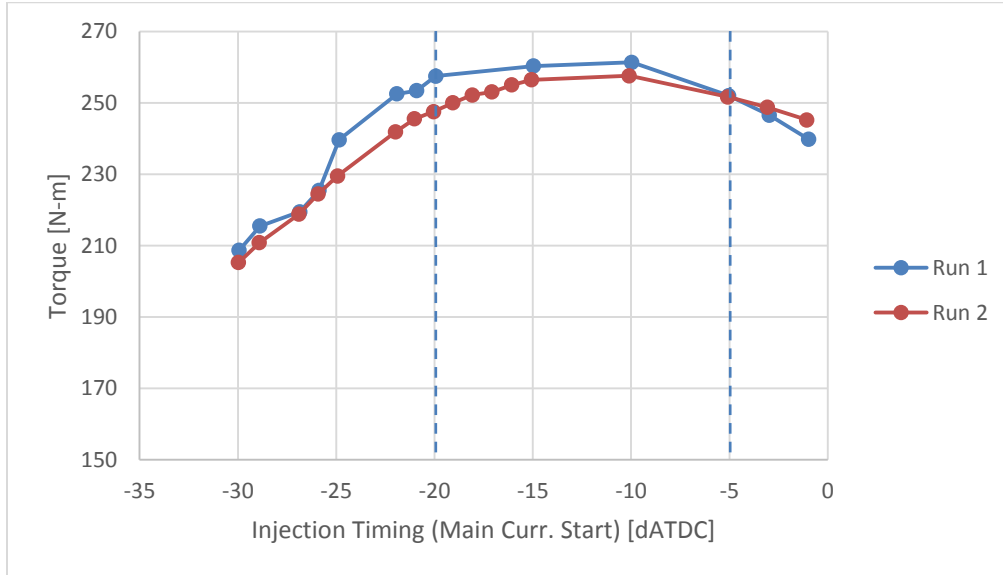


Figure 1: Brake torque (N-m) versus injection timing at high load ($\dot{m}_f = 2.45$ g/s). The additional data points in run 2 between 15 and 20 °BTDC are intended to provide additional resolution in the plateau (bracketed by dashed lines).

The plateau is defined for this study as the injection timing range for which the lowest brake torque is greater than 96% to 97% of the true maximum brake torque recorded. This minimum acceptable torque output would be unlikely to change the utility of an engine in commercial use, since it amounts to about 8 N-m deviation from the true maximum of roughly 260 N-m.

Mass Fraction Burned by the ROHR Integral Method

The mass fraction burned by the premixed and diffusion combustion mechanisms is determined for each injection timing along the plateau using the integral method discussed in Chapter III. Figure 2 shows the heat release rate curves for the injection timings along the torque plateau, each of which was analyzed for premixed and diffusion burn fractions. The curves for injection timings of 5, 10, and 15 °BTDC differ qualitatively from their counterparts in that the second peak for each is higher than the first. This suggests more dominant diffusion burn.

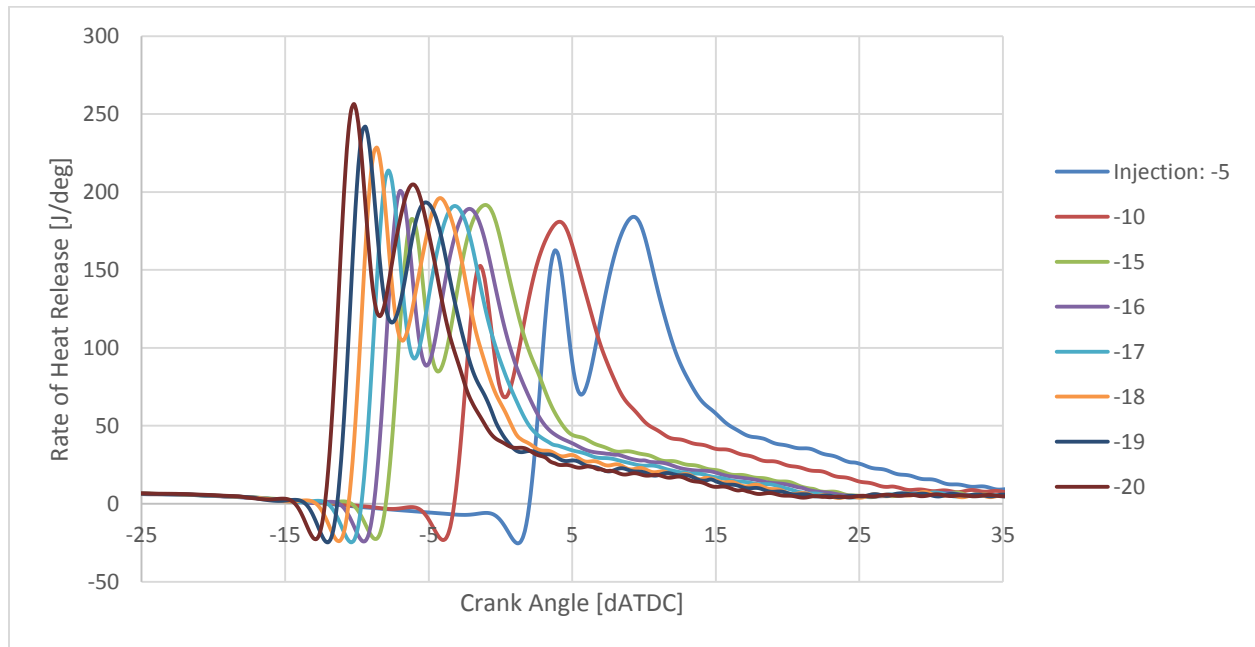


Figure 2: Rate of Heat Release (J/deg) vs. Crank Angle (deg) for the injection timings along the torque plateau. These curves allow integration for mass fraction burned by each mechanism.

A simple trapezoidal integration was applied since the ROHR at each injection timing has a high resolution of 0.2 °CA, such that accuracy remains high. For all timings, the combustion duration was taken to be 50 °CA from the start of injection, since the curves again stagnate near zero ROHR afterwards. From this integration, the mass fractions burned by each type of combustion are shown in Table 3.

Table 3: Tabulated integral approximations of total heat release, premixed burn fraction, and diffusion burn fraction at high load.

Injection Timing (°ATDC)	Total Apparent Heat Release (kJ)	Premixed Burn Fraction	Diffusion Burn Fraction
-20	2.08	0.295	0.705
-19	2.03	0.283	0.717
-18	2.03	0.266	0.734
-17	2.03	0.243	0.757
-16	2.03	0.225	0.775
-15	2.06	0.202	0.798
-10	2.08	0.167	0.833
-5	2.10	0.174	0.826

The integral method with a rather arbitrary 50° duration after the start of injection is shown to provide reasonable consistency by the total apparent heat release (~2.05 kJ). The total heat release is expected to be roughly constant since fuel delivery rate is nominally constant for these tests. Given the known fuel mass flow and net heating value, as well as the calculated combustion efficiency, the error in the integrated apparent heat release is between 5 and 10% for all cases using the nominal 50° combustion duration. Furthermore, the error is consistently an underestimation of the expected apparent heat release, which is important because it confirms

that the simplifications and assumptions do not allow an apparent heat release greater than the theoretical limits imposed by the fuel's heating value.

Since the heat release rate model is a simplification of the real process (for example, the ideal gas assumption is applied to simplify the computation), this accuracy is reasonable, and the integrated premixed and diffusion burn fractions may also be considered reasonably representative of the true proportions of premixed and diffusion flames at each test condition.

As expected, the data also show a decreasing trend in premixed burn fraction as injection occurs closer to TDC. However, the premixed burn fraction is smaller than what was initially hypothesized. The changing emissions spectrum discussed in Chapter I, as well as the emissions data collected in this study suggested that some regimes along the brake torque plateau were dominated by premixed burn. Two likely explanations are as follows.

First, the underestimation of the total apparent heat release may be due to an underestimation of heat release during the premixed part of the cycle, which would in turn result in a reduced apparent premixed burn fraction. For instance, the true premixed burn fraction might be obscured by overestimation of the heat of vaporization in the ROHR approximation. The ROHR curves have a rather deep negative region before premixed burn begins [Figure 2], which is nominally due to the heat absorbed by the liquid fuel as it vaporizes. If the heat of vaporization were overestimated by the approximation, the premixed burn fraction would in turn be reduced, since the apparent start of combustion would be delayed. The true premixed burn fraction at each

injection timing, therefore, may be higher than the fraction integrated along the apparent ROHR curve.

Second, premixed burn may simply produce strong, sensible combustion behaviors even when it does not dominate the MFB spectrum. The initial ROHR peaks in Figure 2 and the emissions profiles shown below demonstrate premixed burn characteristics, even though no computed premixed burn fraction ever constitutes a majority of the total MFB at a given injection timing.

Mass Fraction Burned by Fuel Flow Profile Method

The injection profile model does not illustrate trends as clear as those shown by the common ROHR integral approach. While the premixed burn fraction determined by this method generally decreases as injection is retarded, there are two notable deviations from the trend. Compared to the results from the other injection timings, the earliest injection timing, 20 °BTDC, corresponds to an unexpectedly small calculated premixed burn fraction, and the latest injection timing, 5 °BTDC, corresponds to an unexpectedly large calculated premixed burn fraction [Table 4].

Table 4: Premixed and diffusion burn fractions calculated from the triangular model of the fuel injection rate profile. These fractions are higher at every injection timing than those found by the ROHR integration.

Injection Timing (°ATDC)	Premixed Burn Fraction	Diffusion Burn Fraction
-20	0.392	0.608
-19	0.418	0.582
-18	0.392	0.608
-17	0.367	0.633
-16	0.343	0.657
-15	0.320	0.680
-10	0.276	0.724
-5	0.343	0.657

This method appears to overestimate premixed burn fraction and underestimate diffusion burn fraction, which is understandable in light of the model's assumptions about fuel-air mixing. As discussed in Chapter III, all fuel injected before the start of combustion was assumed to burn in the premixed manner, while all fuel injected after the start of combustion was assumed to burn diffusively. While this assumption simplifies and standardizes the process by which MFB is calculated for each mode, it is necessarily not true in the real engine since the mixing process is time-dependent. For instance, the earliest injected fuel elements might be fully mixed with air before the combustion event, but the fuel elements injected just before the start of combustion would certainly not have time to mix fully within combustible limits before the other premixed fuel ignited the cylinder contents.

The premixed burn fractions from the novel, fuel flow model integration are compared to the premixed burn fractions from the typical ROHR integration in Figure 3. In all cases, despite the two deviations from the trend, the values determined from the fuel flow model are greater than those calculated from the ROHR curves. The true value of the premixed burn fraction is likely to fall between the calculations by ROHR integration, which appears to underestimate premixed burn fraction (see Mass Fraction Burned by the Integral Method above), and the approximation based on a triangular injection model, which appears to overestimate the premixed burn fraction due to the mixing assumptions applied during the integration. Even so, the premixed burn fraction is never the majority of the total MFB, contrary to the hypothesized conditions at the advanced injection timings along the torque plateau.

Although the increase in premixed burn fraction determined by the fuel flow model was unexpected, a similar, smaller increase is also observed in the ROHR integration. The likely explanation is that the phenomenon of retarded premixed combustion is beginning. In other engine research, similar late injection regimes are often applied to help attain LTC since very late timings maximize ignition delay, and accordingly, premixing of fuel and air [6]. In this case, the latest timing along the brake torque plateau caused greater premixing.

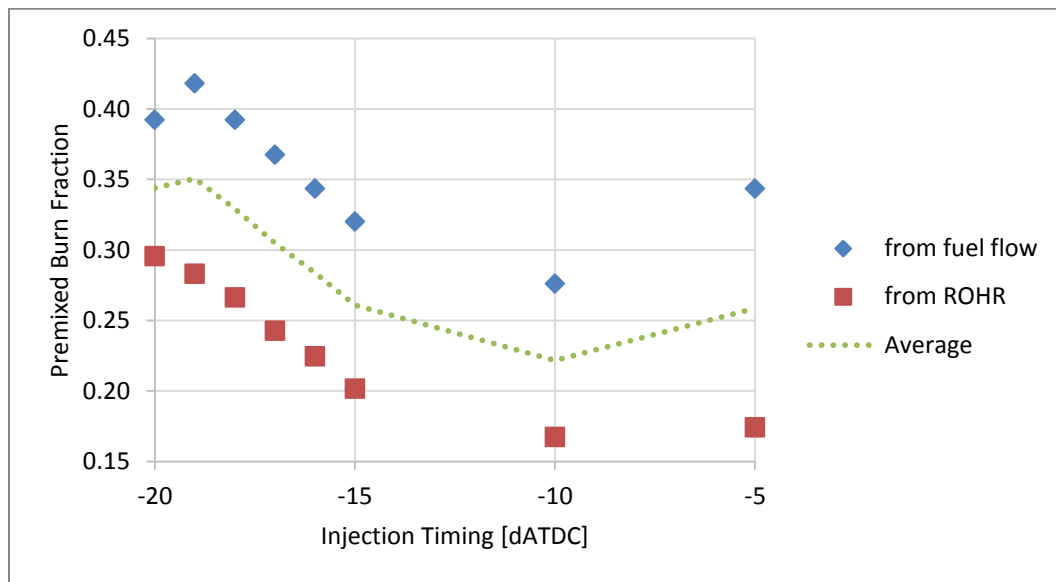


Figure 3: Premixed burn fractions determined by both the triangular fuel flow rate model and the typical ROHR integration. The average is shown to illustrate that the true fractions likely fall in between the values calculated by each method.

The dotted curve illustrating the average of the fractions at each injection timing is included only to highlight the likelihood that the true premixed burn fraction falls between the results of the two approximation methods. It is not intended to illustrate a trend in the premixed burn fractions or to indicate the precise, correct values of the fractions.

Emissions—NO_x and Smoke

Nitric oxide and smoke emissions are important to this study because they hint at the premixed and diffusion burn fractions, and because they link the study to a practical implication—emission reduction. These emissions are plotted together in Figure 4 (note the different scales on the vertical axes). The lines connecting the data points do not suggest trends in and of themselves, but rather demonstrate the sequence of each data set as the two series cross one another.

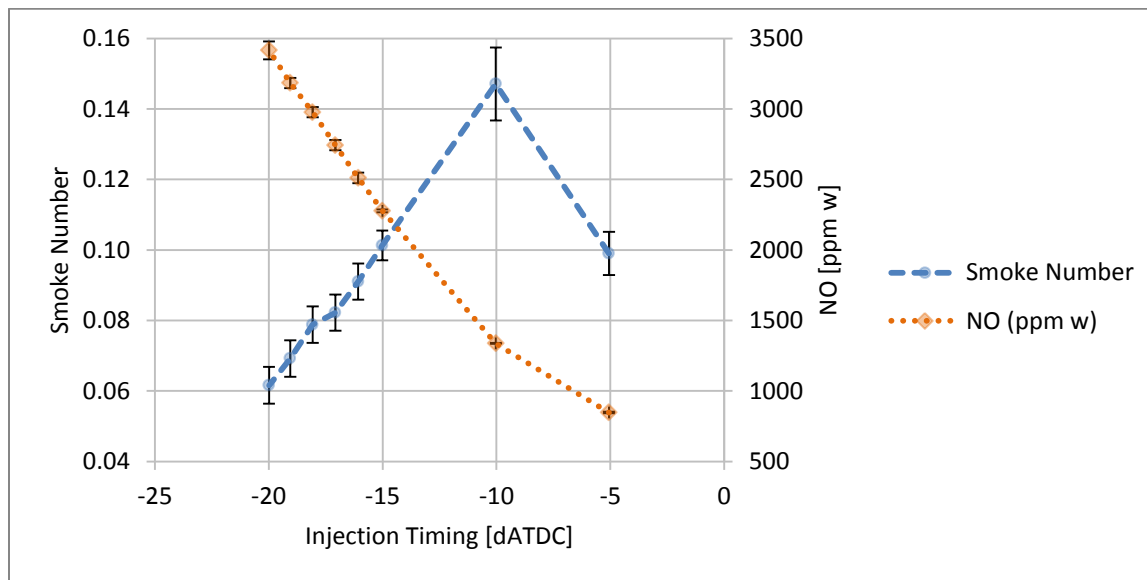


Figure 4: Smoke and NO (ppm w) emissions vs. injection timing. The error bars represent one standard deviation.

Since injection timings from 16 to 19 °BTDC were tested only once, the average standard deviation in smoke number and NO for all other timings with multiple test runs was used to make the error bars on those single-trial timings. The NO concentrations and smoke number values for timings along the torque plateau help verify the trends in premixed and diffusion burn composition shown by the integral method.

Reduced smoke emissions at the latest injection timing (5 °BTDC) appear to contradict the conventional smoke-NO_x tradeoff [6], but when the premixed and diffusion burn fractions are considered, the decline makes more sense. Injection at 10 °BTDC produced the highest diffusion burn fractions by both integration techniques, whereas the latest timing at 5 °BTDC resulted in slightly more premixed burn once more. It appears that in this engine, the injection timing which yields the greatest ignition delay occurs between 5 and 10 °BTDC, such that the ignition delay for injection at 5 °BTDC allows the fuel and air more time to mix before combustion begins [6]. Therefore, a drop in smoke emissions for the latest injection is plausible even though the expected smoke-NO_x tradeoff still applies in general.

In-Cylinder Pressure and Pressure Rise Rate

In-cylinder pressure is an important consideration in engine design and operation because the engine must withstand the high stresses induced by high cylinder pressures. Additionally, the peak pressure and the maximum rate of pressure rise qualitatively show premixed and diffusion burn fractions when two or more operating conditions are compared. Figure 5 shows the pressure variation with time (on a crank-angle basis) for each timing along the plateau.

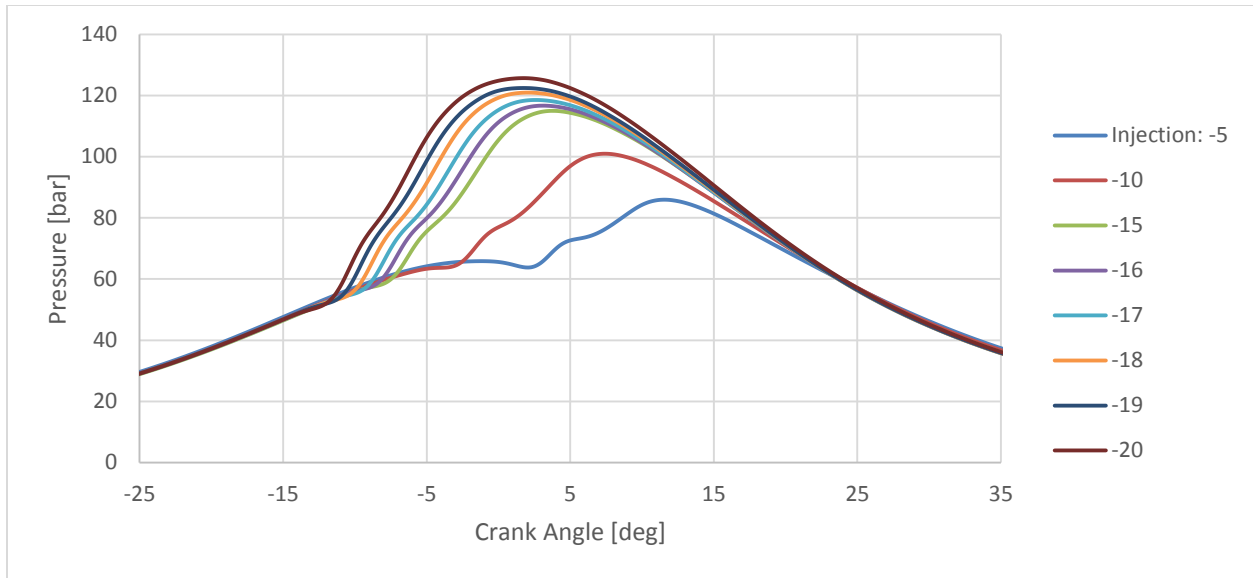


Figure 5: In-cylinder pressure as a function of crank angle for the injection timings along the torque plateau at high load. The profiles for injection timings of -5 and -10 °ATDC are qualitatively very different from the other timings.

In tandem with the peak pressures shown on the graph, the maximum rate of pressure rise decreases as injection timing moves closer to TDC [Figure 6].

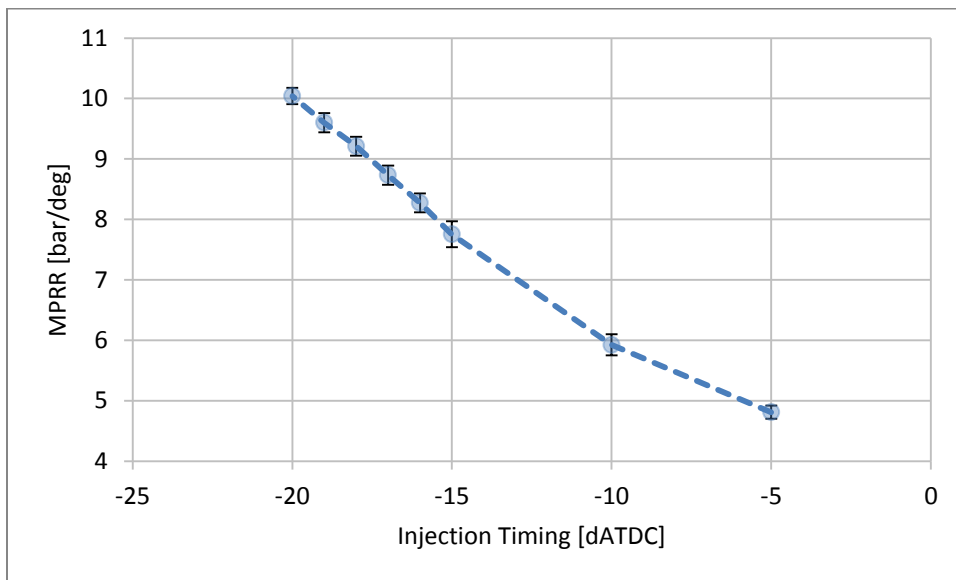


Figure 6: MPRR for each injection timing along the torque plateau at high load. The earliest timings experienced the greatest MPRR.

These pressure and pressure rise data agree with the trends shown in premixed and diffusion burn fractions, revealing decreasing premixed burn fraction as injection timing is retarded.

Efficiency

The fuel conversion efficiency of the engine for each injection timing along the plateau is also an important consideration. Fuel conversion efficiency is defined as the ratio of work per cycle to the amount of fuel energy supplied per cycle or the ratio of engine power to the amount of fuel energy supplied per unit time [1]. For the engine at steady speed with measured torque, engine power is simple to calculate, fuel mass flow rate is also measured, and the fuel's net heating value is known. In Figure 7, the error bars represent the root square sum of the relative uncertainties (percent) in engine power and fuel flow rate, which are the standard deviations of those quantities divided by their averages.

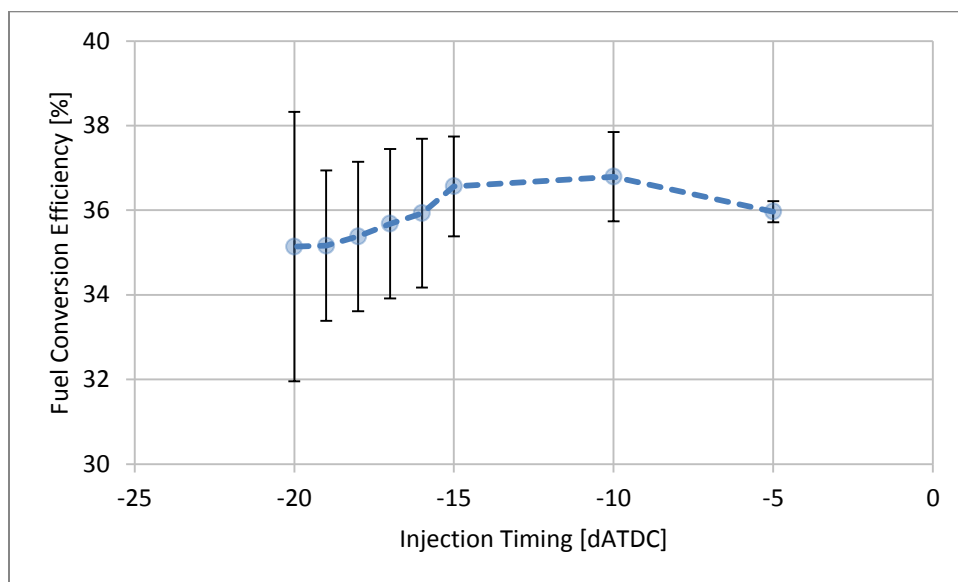


Figure 7: Percent fuel conversion efficiency for each injection timing along the torque plateau. Note the range on the vertical axis from 30 to 40%. The uncertainty is highest for the earliest injection timings.

The fuel conversion efficiency at each injection timing falls within reasonable, typical values for DI diesel engines [1]. Furthermore, the change in fuel conversion efficiency over the range of injection timings in the torque plateau is nearly negligible, with a maximum efficiency of 36.79% at $\theta_{inj} = 10^\circ \text{BTDC}$ and a minimum efficiency of 35.14% at $\theta_{inj} = 20^\circ \text{BTDC}$.

CHAPTER V

CONCLUSION

With the goal of reducing emissions or exploiting an opportunity to lighten production engines, this study quantifies the premixed and diffusion burn fractions present along the brake torque plateau. The data reveal that premixed combustion never dominates the MFB profile as the hypothesis (based on preliminary emissions, MEP, and pressure data from other studies [1,2]) suggested.

However, since the plateau in the present study exhibited similar emissions, ROHR, and pressure characteristics resolved in greater detail since the targeted problem was MFB in the diffusion and premixed modes, the author concludes that so-called premixing-dominated characteristics are simply able to be achieved with non-majority fractions of premixed combustion, since premixed fuel-air mixtures burn so much more intensely than diffusive flame. The presence of these expected emissions and pressure trends means that the hypothetical implications may be realized in engine design and operation.

Relationships Shown by MFB Integrations, Emissions Data, and Pressure Data

Both MFB integration methods, although differing in degree, suggest largely the same trends as those hypothesized. Namely, the premixed burn fraction decreases as injection timing is retarded [Tables 3 and 4]. The discrepancy between the two methods was attributed to possible underestimation of premixed heat release in the ROHR integration and simultaneous, confirmed overestimation of premixed heat release in the fuel injection profile integration.

On the other hand, the emissions and pressure data confirm qualitatively the behaviors discussed in Chapter I [Figures 4, 5, and 6]. These confirmations of anticipated combustion behavior along the torque plateau indicate that premixed burn may dominate the overall characteristics of combustion even when it does not represent a majority of the MFB, which is possible since premixed combustion is much more intense than diffusion burn.

Implications and Practical Findings

Because the hypothesized trends remain apparent even though the premixed burn fractions never clearly dominate the MFB spectrum, the implications presented in Chapter I may be fully explored.

First and foremost, the confirmation of the emissions spectra in terms of smoke and nitrogen oxides provides insight into cleaner operating regimes for engines operating near MBT. Since NO_x emissions are expensive and difficult to trap or decompose [7], later injection timings which discourage premixed combustion are favored from an emissions perspective. Smoke emissions, which increase greatly at moderately retarded timings while NO_x output is significantly reduced, are more favorable from this standpoint because mechanical smoke filtration is a cheaper alternative to chemical filtration of NO_x .

Conveniently, this simple optimization for cheaper emission control along the brake torque plateau complements the best cylinder pressure conditions near MBT. The peak pressures and maximum rates of pressure rise at moderately retarded timings along the torque plateau are

significantly lower as injection timing is retarded towards TDC, which means that peak mechanical stresses on the engine components are reduced in those regimes. Operating at these retarded conditions may allow longer component life or possibly even cheaper, lighter engine design due to the lower peak cyclic stresses.

Finally, these two advantages are not significantly offset by any changes in efficiency along the MBT plateau. The fuel conversion efficiency, indicating the engine's overall ability to convert fuel's energy into work, is nearly constant along the plateau, remaining at about 35%, which is typical for DI diesel engines.

Ultimately, improved engine operation may be achieved at retarded injection timings along the brake torque plateau at high load because of advantages in terms of emissions and mechanical stresses, without substantial costs in terms of efficiency or useful work.

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